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DESIGN OF ANNULAR COMBUSTION CHAMBER FOR A MICRO TURBOFAN ENGINE

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ABSTRACT

The design of an annular combustion chamber for a micro turbofan engine is emphasized in the present work. The combustion chamber is designed using biogas as a fuel. The design principle is based on the constant pressure, enthalpy addition process. The dimensions of the combustor are calculated based on different empirical formulae. The air mass flow is then passed round across the zones of the combustor. The cooling specification is met using the cooling holes. The whole combustion chamber is modelled using suitable modelling software. The model is analyzed using various parameters at various stages and levels to determine the desired design. The aerodynamic flow characteristics are numerically simulated by means of the analysis software. The air-fuel mass ratio, combustion turbulence, the thermal and adequate cooling analysis is carried out. The expected outcome is presented in image outputs and graphs.

KEYWORDS: Design, Annular Combustion Chamber & CFD Analysis

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Original Article

1. INTRODUCTION

The scope of this article is to present a design process for a combustor which will be part of a micro turbofan engine for a cogeneration power plant which works after a Brayton recovery cycle. The compressor and the turbine are the single-stage centrifugal type. The combustion chamber is of annular type. The combustion chamber, or combustor, of a gas turbine, is the device that receives the pressurized air from the compressor and promotes its mixture with the fuel in order to release the heat energy through a combustion reaction.

The turbofan or fanjet is a type of air breathing jet engine in which turbine-driven fan provides additional thrust. Modern turbofans have either a huge one-stage fan or a compact fan with various stages. An early configuration layout integrated a low-pressure turbine and fan in a single rear-fitted unit.

Gas turbines function with a high surplus of air, usually out of the combustibility constraints, and so a flame tube, or liner, is used to enhance the diffusion of air through the reactor. Basically, the liner divides the combustion chamber into three zones: primary zone, secondary or intermediate zone and dilution zone. The main purpose of the primary zone is to moor the flame and allocate adequate time, temperature, and turbulence to attain essentially total combustion of the incoming fuel-air mixture.

2. DESIGN PROCESS

The first step in designing the combustor was the determination of the excess air and fuel mass flow starting from the input data presented in table 1 and table 2.

Table 1: Parameters of the Micro Turbofan Cycle

Parameter	Value	Units
Compression Ratio	5:1	
Intake temperature of Turbine (T3)	1170	K
Output temperature of Compressor(T2)	745	K
Air Mass flow	2.4	kg/s
Efficiency of air compressor	78	%
Efficiency of turbine	82	%

The density of the biogas, according to the chemical composition in Table 2 is 1.31 kg/Nm3. The low calorific power of the biogas was determined using eq. 1:

$$Hi = 12720 \cdot (CO) + 10800 \cdot (H_2) + 35910 \cdot (CH_4) + 23400 \cdot (H_2S),\ kJ/Nm^3 \tag{1}$$

Where, (CO), (H_2), (CH_4) and (H_2S) represent the volume percent for each component of the biogas [10].

Table 2: Chemical Composition of the Fuel

Components used	Volume percent (%)	Density (kg/Nm3)
CH_4	48	0.650
CO_2	43	1.838
N_2	3	1.161
O_2	0.7	1.323
NH_4	0.4	0.7
H_2	0.2	0.085
CO	0.2	1.10
H_2S	0.2	1.429

Thus, a low calorific power 18048 kJ/Nm3 was obtained. Or taking into account the biogas density, a low calorific power stands at14889 kJ/kg. Eq. 2 was used for determining the theoretical quantity of oxygen necessary for complete combustion [10]:

$$O_{min} = 0{,}5.\ [(CO) + (H_2)] + \sum(m + \frac{n}{4}).\ (CmHn) + 1{,}5.\ (H_2S) - (O_2)\ \frac{m_N^3}{m_N^3} \tag{2}$$

where (CO), (H_2), (C_mH_n), (H_2S) and (O_2) represent the volumetric participations for each component of the biogas [5]. Using Eq. 2, for m=1 and n=4, the following value was obtained: O min = 0.994. The theoretical quantity of air necessary for complete combustion was determined using eq. 3:

$$L_{min} = \frac{O_{min}}{0{,}21} \tag{3}$$

obtaining the value: 4.74. The excess of air was determined using eq. 4:

$$\alpha = \frac{H_i - Cpg\ .\ T_3}{Cpg\ .\ T_3\ .\ L_{min} - Cpa\ .\ T_2\ .L_{min}} \tag{4}$$

where C_{pa} represents the specific heat of air and C_{pg} represents the specific heat of exhaust gases [9]. For T_2 =745 K and T_3 = 1170 K an air excess of 5.33 was obtained. For an air mass flow of 2.6 kg/s and using eq. 5:

$$m_c = \frac{m_a}{\alpha \cdot L_{min}} \left[\frac{kg}{s}\right] \quad (5)$$

a fuel mass flow of 0.092449 kg/s was obtained. The next step in designing the combustor was to determine the excess of air and the temperature alongside the liner. For this purpose, the combustor was divided into three regions:

- The fuel injectors region
- The primary zone
- The dilution zone

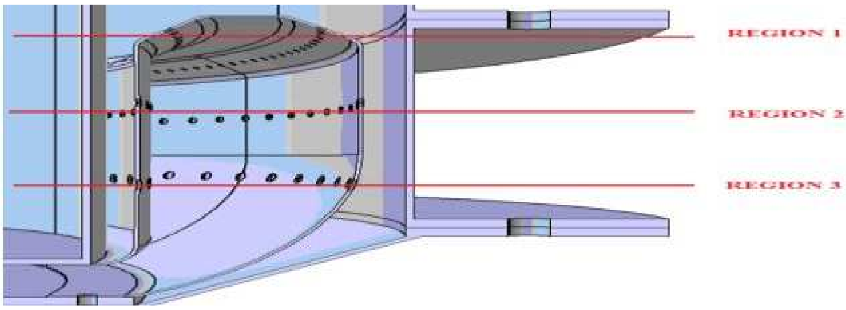

Figure 1: The Three Regions of the Combustion Chamber.

Taking into consideration the information presented in the specialty literature [1], it was considered that 10% of the total air mass flow enters the fuel in the injectors region, 18% of the total air mass flow enters in the primary zone and the rest of 72% enters the dilution zone.

Table 3: Parameters Resulted for the Three Regions of the Combustion Chamber

Regions	1	2	3
m_a(kg/s)	0.260	0.468	1.872
α (excess air)	0.593	1.661	5.932

The temperature along the combustion chamber was determined using eq. 6:

$$T = \frac{efic \cdot H_i + \alpha \cdot L_{min} \cdot T_2 \cdot H_i}{(1+\alpha \cdot L_{min}) \cdot c_{p\_g}} [K] \quad (6)$$

where *efic* represents the combustion efficiency. Its values were taken according to [7]. As expected, the temperature maximum value is obtained in the primary zone.

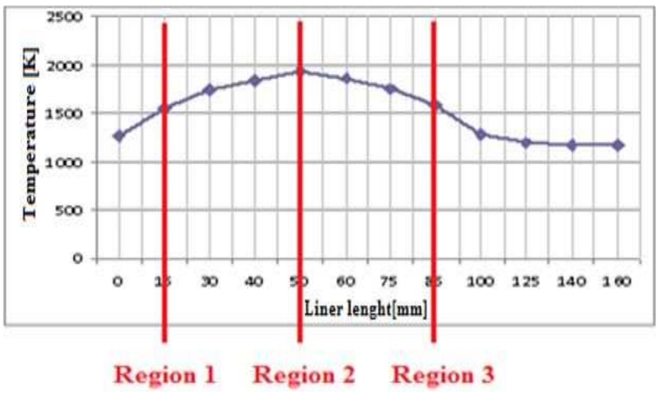

Figure 2: Temperature Distribution.

The final step in the design process was to determine the velocities outside and inside the fire tube and the linear holes diameters based on the jet penetration. The pressure at the compressor's exit is 500000 Pa (p2). Since the distance from the compressor exit to the combustor entrance is considerable due to the micro gas turbine constructive solution, it was considered that the air pressure at the combustor entrance is 475000 Pa.

The velocities outside and inside the fire tube has been calculated using eq. 7:

$$v = \frac{mi}{\rho \cdot S}\left[\frac{m}{s}\right] \tag{7}$$

where S represents the section area.

The jet penetration was calculated using Eq. 8 [6]:

$$H_{ji} = 3.1\,.\phi\,.(0.3 + 0.415\,.\frac{v_a}{v_g}) \tag{8}$$

Table 4: Jet Penetration

Region	1	2	3
Air Velocity($V\ a$)	30.16	27.41	22.14
Fuel Velocity(Vg)	4.13	10.89	22.43
Hole diameter ϕ (mm)	2.5	5	6.5
Jet penetration (Hji)	25.45	20.72	14.25

Based on the calculations presented above, a first version of the combustor geometry, presented in figure 3, has resulted.

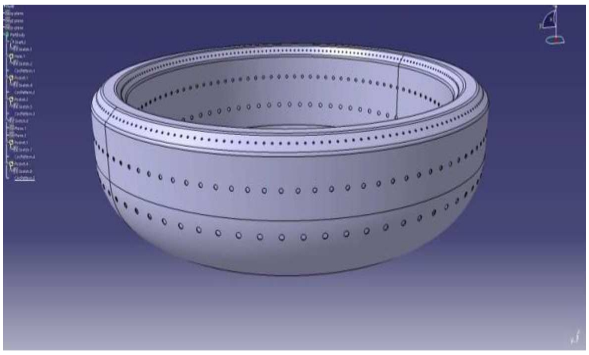

Figure 3: Fire Tube.

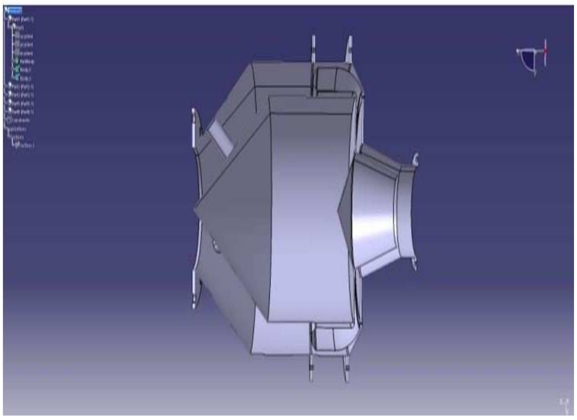

Figure 4: Combustion Chamber Assembly.

3. RESULTS (3-D SIMULATION)

In order to obtain quicker results due computing limitation, the model was simplified into a 22.2-degree cut section for the combustor. The computational aerodynamic analysis is carried out to validate theoretical results and to obtain a detailed preview of the outcome design. The numerical simulations were carried out using ANSYS CFX software. An RANS approach was used with an unstructured type mesh has been generated for the computational domain and with the Domain Motion Stationary. Number of Tetrahedral Elements: 8231702.

The following boundary conditions were used.

- At air inlet, there were imposed the air mass flow and temperature,
- at fuel inlet there were imposed the fuel mass flow and temperature and
- at the outlet, the pressure was imposed.

The Eddy Dissipation combustion model was used, in combination with the K-epsilon turbulence model. This model was chosen because it allows accurate simulation of the heat release and the distribution of the main chemical species.

The velocity distribution presented in figure 5 shows high velocity in the central region. Jet penetration is very strong, thus the created turbulence will affect combustion process.

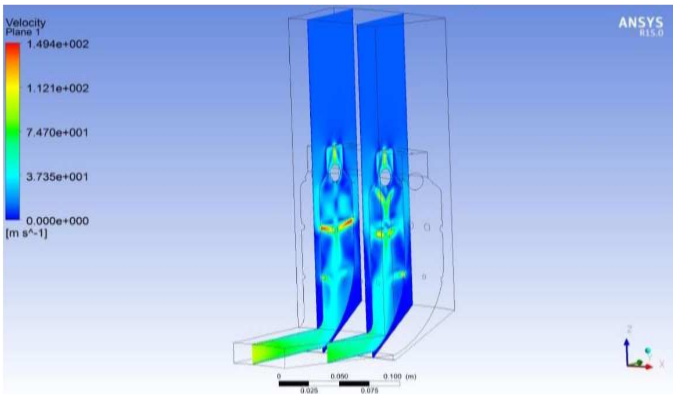

Figure 5: Velocity Distribution.

From figure 6, it can be observed that the flame temperature presents high values mainly near the combustor's walls. This is in good correlation with the velocity profile presented in figure 5. The high-velocity values from the central region of the fire tube makes mixing of the air and fuel difficult in the primary region.

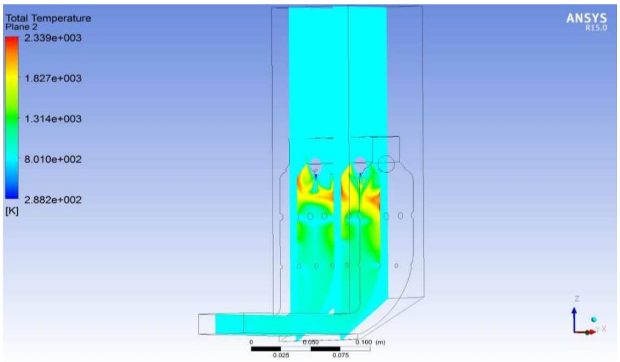

Figure 6: Total Temperature Distribution.

Figure 7, = are presented with 4 temperature isosurfaces inside the fire tube (red = 2300 K, orange = 2100 K, yellow = 1700 K, green 1300 K). As it was observed before, the high flame temperatures are developing near the walls. This is not a good sign. It can lead to serious damage to the fire tube. It also can be seen that the flame has a very irregular structure.

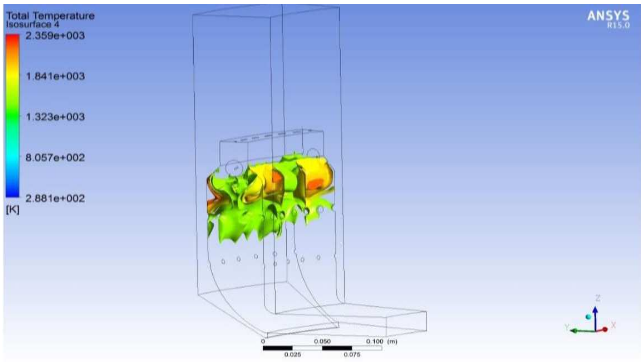

Figure 7: Total Temperature Isosurface Distribution.

The average temperature at the exit of the combustor was 1160K. Even though, this temperature value is very close to the one imposed in table 1, based on the results obtained so far, it was concluded that same changes have to be done to the combustor geometry. A deflector was added to the original geometry in order to concentrate the flow in the central region of the fire tube and to prevent flame adhesion to the walls of the combustor. The improvement can be observed in figure 8 and figure 9. The flow velocity has diminished and the flame is concentrated in the center of the fire tube.

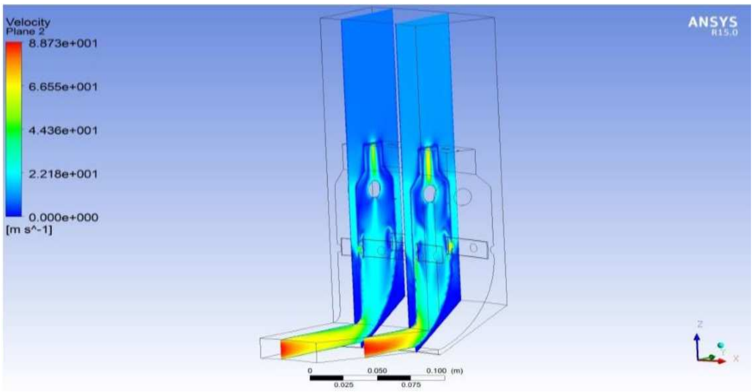

Figure 8: Velocity Distribution.

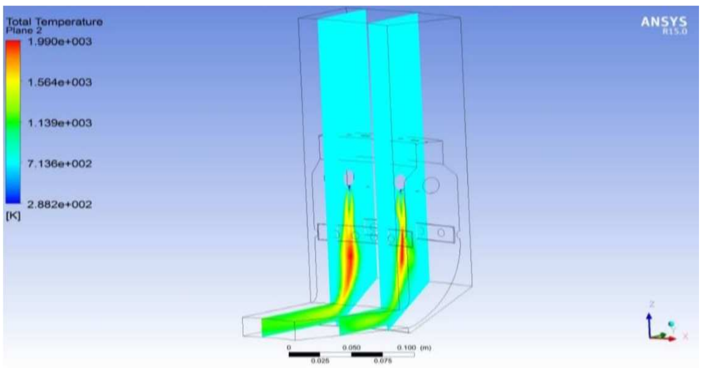

Figure 9: Total Temperature Distribution.

4. CONCLUSIONS

The complete annular combustor design for a micro turbofan engine using just the initial design parameters has been discussed in this paper. This is a design process which can be used for the preliminary design. The transparent and detailed approach is focused on reducing design time and complexity. This gives an overall advantage in total design time and prototype building. Using the process, a practical design is presented. The obtained values are used for modelling and further simplified for analysis. The analysis was also carried out with higher accuracy using the combustion-turbulence interaction model and the results show that the optimum gas exit temperature was obtained for the present design. The design was successfully calculated and modelled.

5. FUTURE SCOPE

The present work can be carried out for unsteady cases. The present work can be extended to know internal behaviour of the fuel and air mixture for various ratios.

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